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CRITERIA FOR FASTENER SYSTEM DESIGN

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Aerospace Corporation
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ABSTRACT

Criteria for fastener system design are suggested. The application of system analysis to mated parts held together with a high-strength fastener is discussed. The torque-tension relationship of each system should be measured, and fracture mechanics should be used to index environmental compatibility. This approach can also be used to produce a cost-effective fastener system design.

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CRITERIA FOR FASTENER SYSTEM DESIGN

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FORWORD

This report is published by Aerospace Corporation, El Segundo, California, under Air Force Contract No.F04695-67-C-0158.

This report, which documents research carried out from July 1966 through June 1967, was submitted for review and approval to Captain William D. Bryden, Jr., on 26 October 1967.

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Approved

W C Riley

W. C. Riley, Director
Materials Sciences Laboratory
Laboratory Operations

Publication of this report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

William D. Bryden

William D. Bryden, Jr., Capt., USAF
Project Officer

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I. INTRODUCTION

Applying system analysis to mated parts that are held together with a fastener is becoming essential as minimum-weight designs call for higher strength materials. The performance of the individual components of a fastener system (i.e., bolt, washer, spacer, nut, lubricant, and protective coating) must be related to the performance of the integrated system (i.e., optimized torque-tension relationships with environmental compatibility). An immediate advantage of optimizing the performance of the fastener system is that some of the stringent requirements imposed on individual components might possibly be relaxed. It has been recommended (Ref. 1) that the fastener manufacturer must be responsible for providing a completely integrated fastener system, but the designer must still be responsible for selecting the fastener system for a specific application.

The performance of the entire unit, i.e., the fastener system, is of concern in this presentation. The serious need for applying systems analysis to the design of mated parts that require a fastener will be demonstrated. The complexities of such an approach will be discussed, and techniques that index the performance of a fastener system will be described. These techniques can be combined with economic considerations to produce a cost-effective fastener system design.

II. NEED FOR EVALUATING TOTAL PERFORMANCE OF A FASTENER SYSTEM

Figure 1 is a simplified illustration of a fastener application. The parts are held together by a bolt inserted through holes in the flanges. Before the nut is put on, various types of washers (e.g., a load-indicating washer) are usually added to the system. Thread design of the nut and the bolt consistent with maximized resistance to fatigue has in all probability been provided by the manufacturer. The design of the head is usually given the same consideration. Each of the components may be of a different material, the nut and bolt might be plated, and lubricants might be added. A further complication is that the flange might be of still another type of metal. The entire unit is assembled and then tightened with a torque wrench to produce a given clamping force, i.e., tensile load.

If the performance of a fastener is not evaluated as an integral system with the mated parts, critical operational problems may be encountered. The measure of the performance of a fastener is its ability to produce a specified torque-tension relationship without degradation due to a lack of environmental compatibility. In one specific case, a valve assembly that had a bolt and nut arrangement similar to that shown in Fig. 1 failed because of a lack of environmental compatibility. When the 17-4 PH stainless bolts were installed with an established torque specification, their contact with the aluminum mating parts caused failures to occur within 5 hr (Ref. 2). The bolt performance was seriously degraded when the bolt became an integral part of a fastener system. The problem might have been avoided had system analysis been applied to the design of the valve and fastener.

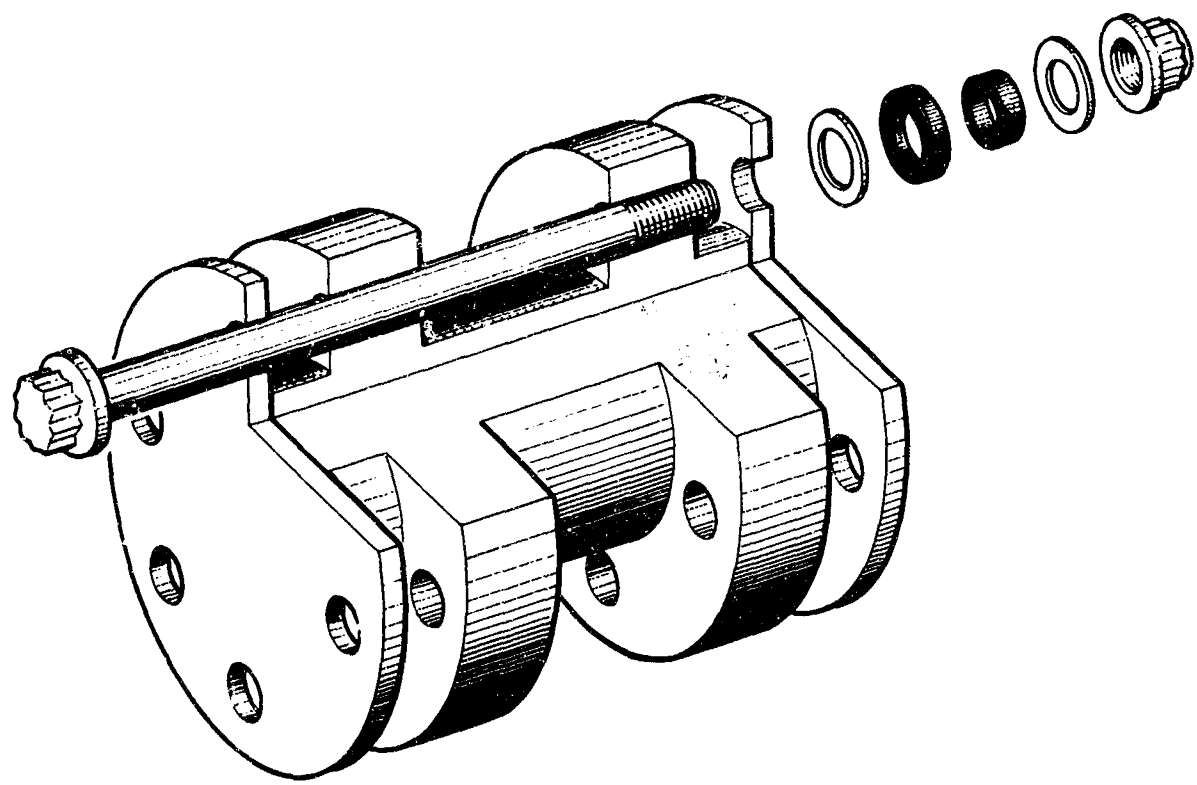


Figure 1. Simplified Fastener System

III. COMPLEXITY OF FASTENER SYSTEM ANALYSIS

The complexity of a fastener system analysis arises from the numerous variables that make up such a system and consideration of their interaction and overall system effects in an operational environment. Definition of the interrelation between coatings and platings for lubrication and for corrosion protection and the resulting torque-load (T-P) relationship can be used to establish the optimum performance of a specific fastener system for a specific application.

A. USE OF COATINGS AND PLATINGS

Platings and coatings may be used for various purposes (Table I). In general, their use in fastener systems is either as a lubricant that minimizes the coefficient of friction or as a protective coating that provides environmental compatibility and protects the fastener against chemical degradation in its operational environment. In aerospace applications, this environment could be long-time exposure in a marine atmosphere or short-time exposure to propellants and fumes.

B. DEFINITION OF ENVIRONMENTAL COMPATABILITY

Fracture of a stressed component in a chemically aggressive atmosphere usually involves (1) an incubation period during which a pit develops that leads to the formation of a crack and (2) a period during which this crack grows to a critical size. Figure 2 shows the relationship of the direction of the applied stress to the direction of the propagating crack. This failure mechanism, defined as stress-corrosion cracking (SCC), becomes more pronounced as

TABLE I. Use of Metallic Platings

Plating	Use	Comments
Silver	Lubricant	High-temperature applications on corrosion resistant fasteners (AMS 2410)
Cadmium	Environmental compatibility	Oxidation resistance on alloy steel bolts to 450°F (Spec. QQ-P-416a, Type II or MIL-C-8837 Type I)
Cadmium	Environmental compatibility	Prevents galvanic corrosion on A-286, stainless steel, and carbon or low-alloy steel bolts in aluminum structures (Spec. QQ-416a, Type I or II)
Chromium	Environmental compatibility	Salt-water protection
Diffused Ni-Cd	Environmental compatibility	Oxidation resistance on alloy steels up to 900°F (Spec. AMS 2416)
Electroplated Ni	Environmental compatibility	Corrosion and oxidation resistance

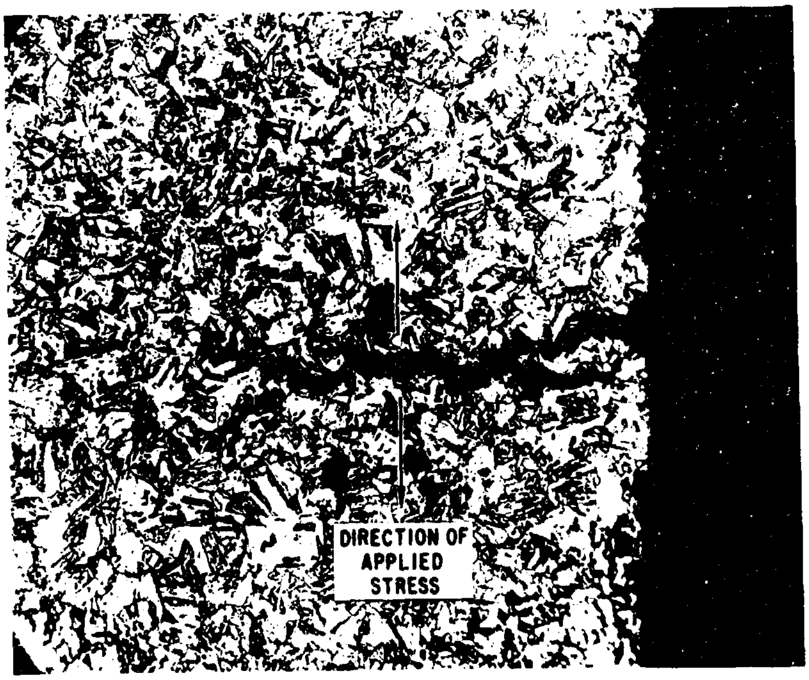


Figure 2. Intergranular Stress-Corrosion Crack Propagation

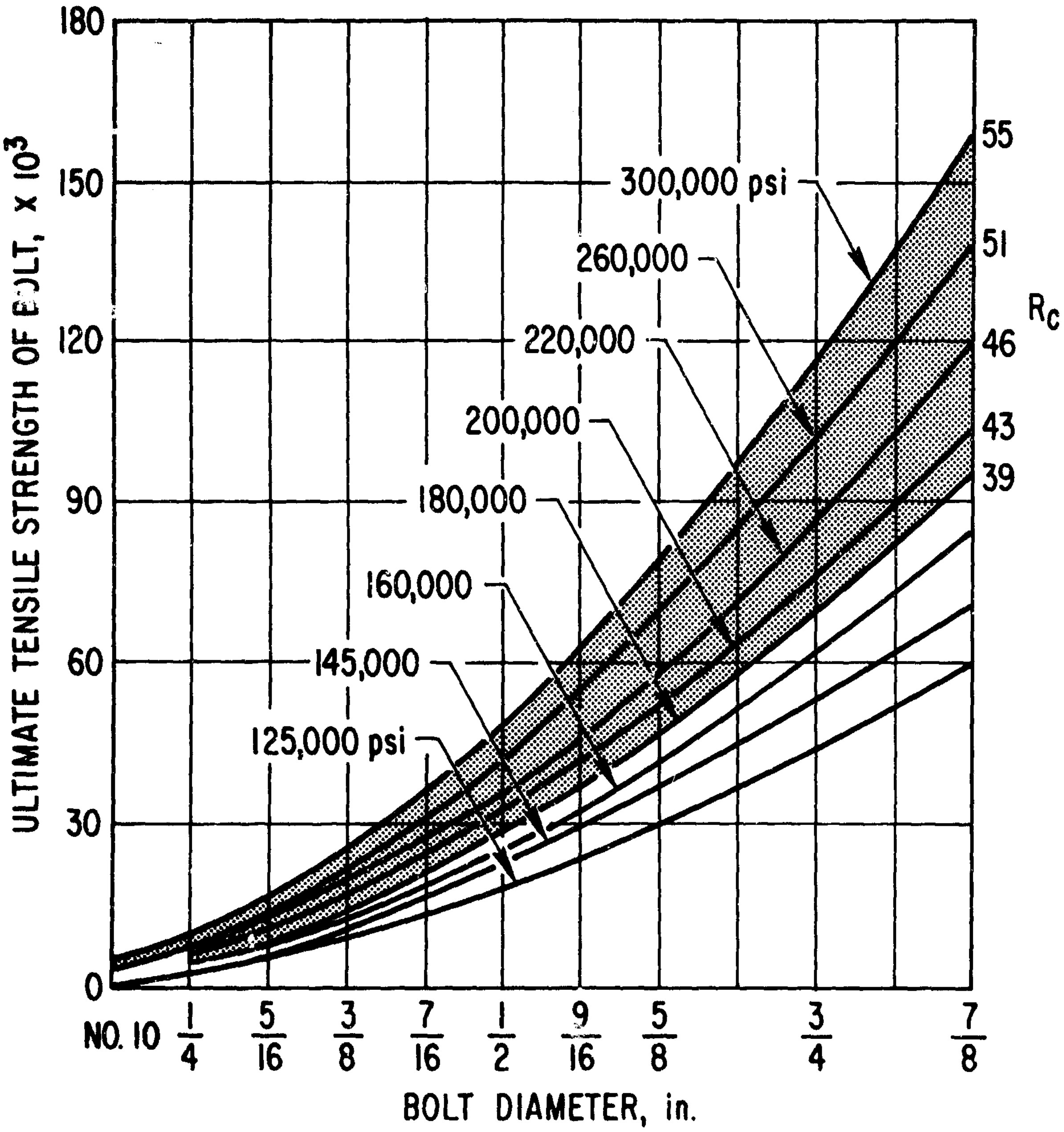


Figure 3. Shaded Area Represents Bolt Conditions Where SCC and HSC Concurrently Exist as Failure Mechanisms in Martensitic Steels (After Ref. 1)

the strength-to-density ratio of the material exceeds 500,000 in. For martensitic steels, the shaded area in Fig. 3 represents bolt conditions where both SCC and hydrogen stress cracking (HSC) are simultaneously operating failure mechanisms.

Despite extensive research on SCC phenomena, the mechanism is not completely understood. In SCC, failure occurs when corrosion, often imperceptible, takes place on a metal loaded in tension. Residual (tensile) stresses may also cause failure. The susceptibility of steels to SCC is affected by the composition and heat treatment of the steel. The severity of the stress (expressed as percentage of the yield point) and the aggressiveness of the environment strongly influence the time to failure.

High-strength bolts loaded in tension can also be subjected to delayed fracture from HSC. These failures occur below the yield stress and have been related to the hydrogen concentrations in the steel. It has been reported that hydrogen concentrations as low as 1 ppm will cause ferrous alloys to crack. HSC is affected by the amount and nature of the hydrogen present, the magnitude and distribution of the internal stresses, and the ability of the material to withstand these stresses. As stress increases, less hydrogen is required for time to failure. It is, therefore, important to keep steel hydrogen-free. Hydrogen can be introduced during melting, heat treatment, pickling (cleaning), or improper electroplating operations. The effects of hydrogen introduced during plating of high-strength steel bolts have generally

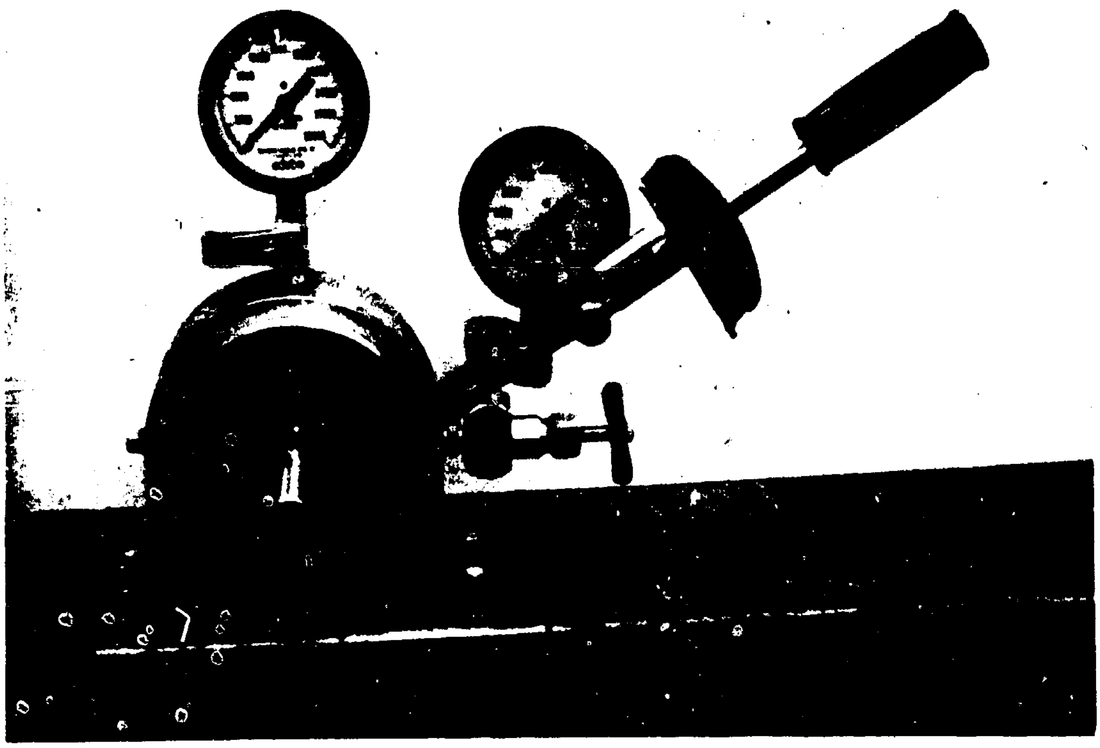


Figure 4. Skidmore-Wilhelm Testing Machine
for Torque-Tension Measurements
of Fastener System

been given adequate consideration; the effects of hydrogen introduced under service conditions have been severely neglected.

The significant difference between HSC and SCC is that HSC is more apt to occur when the stressed component is cathodic, and SCC, when the stressed component is anodic. Therefore, regardless of the type of the metallic coating or plating selected, a potentially hazardous failure situation exists because accelerated failure will occur by HSC, if the coating or plating metal is anodic, or by SCC, if the coating is cathodic.

C. SENSITIVITY OF T-P RELATIONSHIP TO MATERIALS

Once the components of a fastener system are selected, the performance of the entire system must be evaluated with respect to the torque-tensile load (T-P) relationship. The T-P relationship is experimentally determined (Fig. 4), and then the maximum combined stress (σ) at the threads is theoretically calculated (Ref. 3). Some aerospace companies have already developed design specifications that include torque-stress-load relationships for various bolt sizes, thread and head designs, lubricants, coatings, and mating parts. Corresponding materials and heat treatments are also specified.

Table II lists typical data for a fastener system that includes an AN-type, external hex-head, steel bolt, a 347 crescent tapped stud, and an

A-286 washer. From Table II, if a 3500-lb clamping force is required, the maximum combined stress at the threads for a low coefficient of friction (f=0.04) would be 112,000 psi and for a higher coefficient of friction (0.40), the stress would be almost double, or 220,000 psi. Selection of a steel to withstand 220,000 psi introduces many more environmental compatibility problems than the selection of a steel to withstand 112,000 psi. In this particular case, the use of a good lubricant would significantly relax the material requirements. Similarly, the material selected for a washer and nut or the configuration of the bearing surface under the head of the bolt could also significantly alter these requirements.

TABLE II. Torque-Stress-Load Relationships For 1/4-28 Bolt [a]

Lubricant	f	σ/T	P/T	σ/P
MoS_2	0.04	1803	56	32
None	0.40	512	8	63

[a] f=coefficient of friction

σ =maximum combined stress

P=clamping force or axial tensile load, lb

T=torque, in.-lb

D. RETORQUING EFFECTS ON T-P RELATIONSHIP

The most direct method of maintaining a specified clamping force is to check the torque periodically. If vibrations have relaxed the torque, the bolt is retorqued to specifications. In a space vehicle, several retorquing cycles may be necessary prior to launch, and this procedure can change the T-P relationship of the fastener system. Figure 5 shows retorquing data taken from an actual fastener system. Without a lubricant, the clamping force is seen to decrease after four cycles by a factor of 2 (from 10,000 to 5000 lb). With Lox-Safe as a lubricant (a graphite suspension in grease that can be used at liquid oxygen temperatures), the initial clamping force is only 1000 lb greater for the initial torque of 150 ft-lb. After four retorquing cycles, the clamping force increases from 11,000 to 14,000 lb, which is 3 times greater than the unlubricated bolt. Figure 6 compares the bearing surfaces under the head of the bolt. Galling, which occurred in the absence of a lubricant, caused the coefficient of friction to increase with subsequent torquing cycles. Thus, for the same initial torque, the axial load transmitted by the bolt changed significantly. The effect on the maximum combined stress of increasing the coefficient of friction and maintaining a constant torque can be explained from the data in Table I. As the coefficient of friction increases from 0.04 to 0.40, the combined stress decreases by a factor of 3; simultaneously, the clamping force decreases by a factor of 7. Thus, a major portion of the torque energy is dissipated in shear.

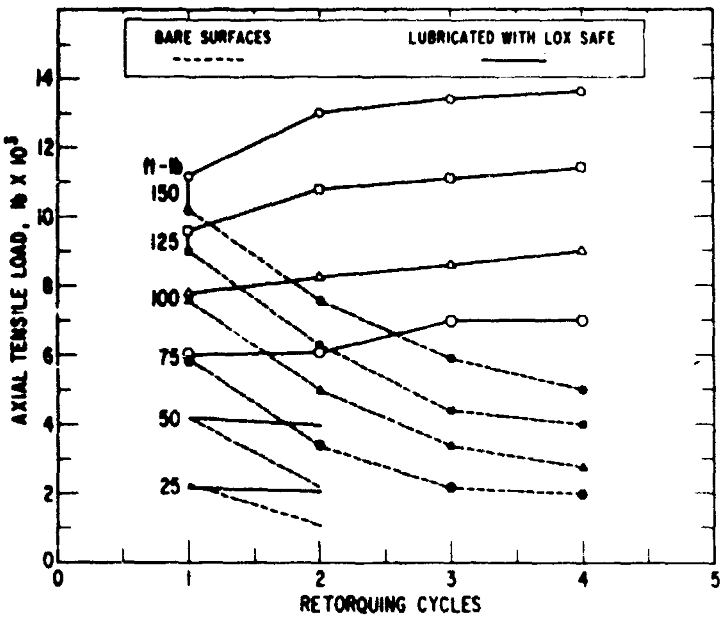
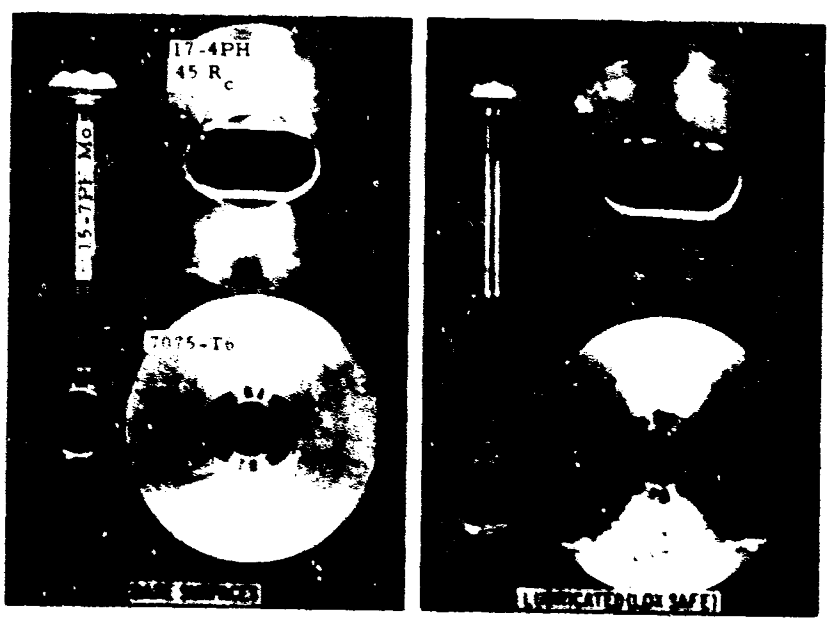


Figure 5. Effect of Lubrication on Torque-Tension Measurements of Fastener System

Figure 6. Galling on Slotted 17-4PH Bearing Surface of a 15-7PH Bolt



IV. TECHNIQUES FOR INDEXING PERFORMANCE

Service failures of high-strength steel bolts inevitably occur in the bolt shank and not in the threads where the combined stress is a maximum value. This failure location is unexpected because, from a comparison of the areas, the stress in the shank is not the point of maximum stress. Figure 7 shows a bar that was V-notched and tested for resistance to HSC. Failure did not occur at the V-notch where the stress was 260,000 psi, but at a location where a pit developed and then grew to a critical size at a stress level of only 130,000 psi. These results indicate the sensitivity of failure mechanisms such as HSC and SCC to notch acuity. The sharper the notch, the lower the threshold stress. The stress dependence of the sharpest notch (fatigue crack) in a specified environment can be analyzed using fracture mechanics.

Data have been recently generated (Ref. 5) on the performance of various coatings and platings of 1/4-28 H 11 steel bolt under SCC conditions. These results show a threshold load of 3500 lb below which no failures occurred in a 3.5% salt-solution immersion test. A typical cross section of a bolt that failed the test is shown in Fig. 8.

Crack penetration under SCC conditions can be measured. From fracture mechanics, where a is the crack depth and 2c is the crack length,

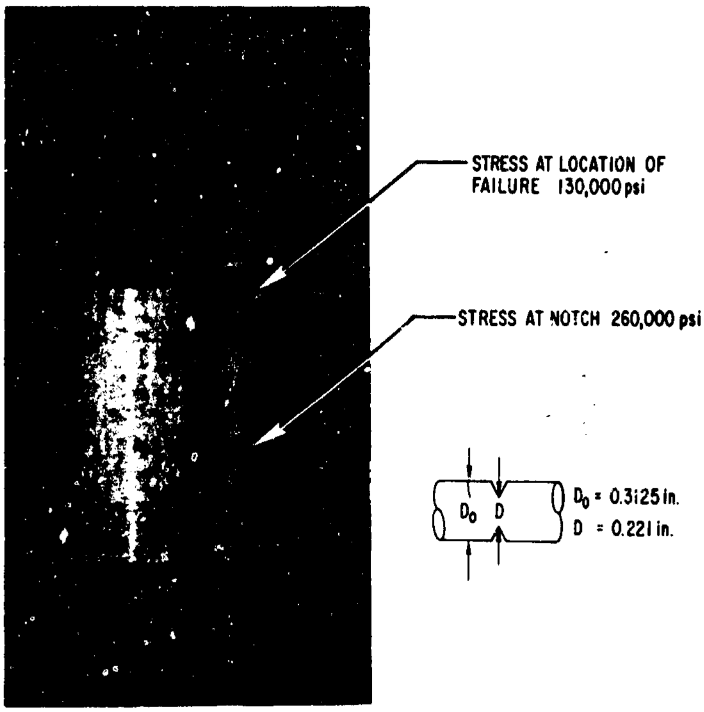


Figure 7. Effect of Notch Acuity on Severity of SSC Attack on Dissimilar Metal Couple (Al/SS) in a Marine Environment

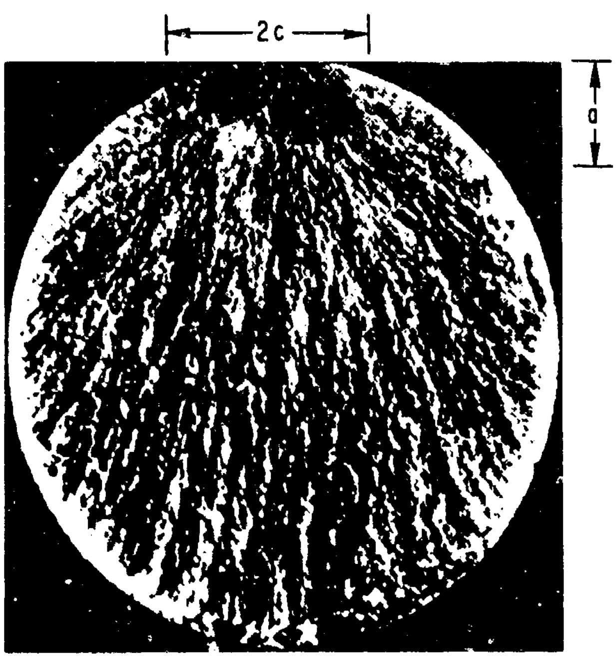


Figure 8. Typical Appearance of a Cross Section of a Bolt that Failed from SCC

$$K_{Ic}^2 = (1.21\pi)(\sigma^2)(a/Q)$$

and

$$K_{Ic} = 30{,}000 \text{ psi-in.}^{1/2} \text{ for H 11}$$

From Fig. 8, $a/2c = 0.4$, and therefore $Q = 2$ and $a/Q = 0.33$ in. (Ref. 5)

or

$$\sigma^2 = \frac{(30{,}000 \text{ psi - in.}^{1/2})^2}{(1.21\pi)\ (0.33 \text{ in.})} = 71.8 \times 10^8$$

$$\sigma = 85{,}000 \text{ psi (load/area)}$$

$$P = 85{,}000 \times 0.0491 = 4{,}200 \text{ lb}$$

This 4200-lb load is just above the failure threshold load of 3500 lb under which crack growth does occur in the 3.5% salt-solution immersion test.

Fracture mechanics can also be used to index crack growth characteristics through the stress intensity factor K_{Iscc} that is some fraction of the K_{Ic}, below which no crack growth occurs. The K_{Iscc}-to-K_{Ic} ratio can be as low as 0.1 for 4340 in a marine environment, which makes this a very poor material for a high-strength steel fastener in such an environment. In fact, materials and environments that have K_{Iscc}-to-K_{Ic} ratios below 0.8 should be avoided for high-strength applications. This approach is limited in that it measures crack growth sensitivity in the propagation step of SCC; i.e., it does not include an evaluation of the crack initiation step. Thus, the most severe conditions are evaluated. If the pit or crack never forms because

the incubation time for crack initiation is longer than the operational time, fracture is not encountered. The same arguments are valid for analyzing plating and coating performance. Ideally, they should offer adequate protection, but improper handling procedures during inspection can prematurely lead to the crack propagation step and to a very rapid failure.

Because high-strength materials, and especially martensitic steels, are susceptible to both HSC and SCC, it is imperative to consider protection against both mechanisms in selecting a plating or coating. The first approach is to protect against the most critical failure mechanism. This requires a thorough understanding of the corrosion current generated by galvanic action. Another approach would be to use an insulating material to prevent a galvanic coupling, oftentimes in the form of a grease such as Lox-Safe, or to remove the possibility of collecting moisture by using a filler material, which again might be a grease or a plastic. To suggest the elimination of all dissimilar metal couples for metallic materials heat treated to a strength-to-density ratio of 500,000 in. would be impractical.

Cathodic protection could be used for metallic materials that are not susceptible to HSC. In general, this would necessitate a face-centered, cubic crystallic structure. Such is the case with austenitic stainless steels or aluminum and its alloys. Hexagonal, close-packed, metallic materials, such as titanium, and body-centered cubic or tetragonal metallic materials,

such as martensitic steels, should be avoided. Thus, the high strength A-286 and Inco 718 fasteners are becoming extremely popular. These alloys are presently limited to maximum strengths of 250,000 psi or 850,000 in. strength-to-density ratios. Ultrahigh-strength steel in the 1,000,000 in. strength-to-density ratios are of the martensitic variety and display susceptibility to both failure mechanisms. Unquestionably, performance evaluation of an entire fastener system in its operational enviornment will become more critical as the strengths of fastener materials increase.

V. COST OPTIMIZATION

Optimum designs that produce a cost-effective fastener system can be obtained by indexing the system performance under the original loading conditions and under a specified environment for a specified time. Fracture mechanics provides the technique necessary for such an analysis.

Crack size and growth under various stresses and environments can be indexed. These parameters can be related to economics by considering the relative costs of inspection for a specific crack size, the cost of the materials, and the cost of the entire fastener system. These analyses may show that the weight or cost of the components of a fastener system should not be minimal and that their reliability need not be the maximum attainable through research and development. The optimum for each parameter should be a function of its contribution to the system. Minimum-weight design criteria do not necessarily yield the most cost-effective fastener system, since the required performance may not be fully defined. Through the fracture mechanics approach, intuitive techniques of cost-optimization can be replaced by analytical techniques.

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VI. SUMMARY AND CONCLUSIONS

1. Fastener system components should be identified with respect to design, material, heat treatment, lubrication, coatings, and platings.

2. The torque-tension relationship for each fastener system should be measured.

3. The entire fastener system should be simulated and exposed to environmental tests.

4. For applications that require fastener material strength-to-density ratios in excess of 500,000 in., more attention should be given to metals and alloys that are not susceptible to hydrogen embrittlement.

5. Ideally, platings and coatings provide adequate protection from SCC and HSC, but a potential fracture hazard always exists. Poor handling procedures during installation, maintenance, or retorquing operations may gall these coatings and thereby rapidly activate a failure mechanism.

6. Fracture mechanics should be used to evaluate environmental (i.e., operational) performance of materials. The basic parameters can be related to costs for material, inspection (NDT), maintenance, and design and can provide a fundamental basis for arriving at a cost-effective fastener system.

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13. ABSTRACT

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14. KEY WORDS

Cathodic Protection
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Environmental Compatibility
Fastener System Criteria
Fracture Mechanics
High-Strength Fasteners
Hydrogen-Stress Cracking
Stress-Corrosion Cracking
T-P Relationship

Abstract (Continued)